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USAAVLABS TECHNICAL REPORT 69-10A

ADVANCEMENT OF SMALL GAS TURBINE COMPONENT TECHNOLOGY, ADVANCED SMALL AXIAL COMPRESSOR (U)

VOLUME I - ADDENDUM
ANALYSIS AND DESIGN

By

James V. Davis

December 1969

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-296(T)

CONTINENTAL AVIATION AND ENGINEERING CORPORATION
DETROIT, MICHIGAN

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FORT EUSTIS, VIRGINIA 23604

(U) The research described herein, which was conducted by Continental Aviation and Engineering Corporation, was performed under U. S. Army Contract DA 44-177-AMC-296(T). The work was performed under the technical management of Mr. David B. Cale, Propulsion Division, U.S. Army Aviation Materiel Laboratories.

(U) Appropriate technical personnel of this Command have reviewed this report and concur with the conclusions contained herein.

(U) The findings and recommendations outlined herein were considered in planning the subsequent phases of the program.

(U) This document is the classified addendum to USAAVLABS Technical Report 69-10A.

Task 1G162203D14413
Contract DA 44-177-AMC-296(T)
USAAVLABS Technical Report 69-10A
December 1969

ADVANCEMENT OF SMALL GAS TURBINE COMPONENT TECHNOLOGY, ADVANCED SMALL AXIAL COMPRESSOR (U)

VOLUME I - ADDENDUM ANALYSIS AND DESIGN

Continental Report No. 1033

By

James V. Davis

Prepared By

Continental Aviation and Engineering Corporation
Detroit, Michigan

for

U.S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

(U) SUMMARY

A complete two-stage aerodynamic axial compressor design was prepared using the optimum axial compressor characteristics obtained from a preliminary design study of a family of advanced axial compressors capable of being matched with the USAAVLABS advanced centrifugal technology. Fully variable part span inlet guide vanes were designed to ensure axial-centrifugal operation capability from 50 to 100 percent of engine speed. Two transition ducts, which differed primarily in length, were designed to provide an efficient aerodynamic flow path from the axial compressor exit to the centrifugal compressor inlet.

The axial compressor has the following aerodynamic design point performance goals:

Airflow = 5 lb/sec
Pressure ratio = 3.0:1
Adiabatic efficiency = 83 percent
Inlet hub tip ratio = 0.494
First rotor tip speed = 1410 ft/sec

(U) FOREWORD

This report, prepared by Continental Aviation and Engineering Corporation, presents Phase I of a three-phase small axial compressor program for the advancement of small gas turbine component technology.

The program was sponsored by the United States Army Aviation Materiel Laboratories under Contract DA 44-177-AMC-296(T), Task 1G162203 D14413.

Phase I presents a study of a family of advanced axial compressors. It is reported as Volume I with an addendum under a separate cover. The addendum is the analysis and design.

Phases II and III are presented in Volume II. Phase II presents the axial compressor fabrication and test. Phase III presents the axial compressor redesign, fabrication, and test.

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(C) COMPRESSOR AERODYNAMIC DESIGN (U)

(C) DESIGN PROCEDURE (U)

The advanced axial compressor was designed on Continental computer program 08.047, Axial Flow Compressor Calculation. This program predicts axial-flow compressor performance by solving the continuity, energy, and complete radial equilibrium equations preceding the following blade rows. The complete radial equilibrium equation used for this compressor design is as follows:

$$\frac{1}{p} \times \frac{\partial P}{\partial R} = \frac{c_u^2}{R} - C_m^2 \times \left[\frac{\cos \theta}{R_c} - \frac{\sin \theta}{C_m} \times \frac{\partial C_m}{\partial m} \right]$$

where:

$\frac{\partial P}{\partial R}$ = radial static pressure gradient, pound per foot squared per foot

c_u = tangential velocity, feet per second

p = mass density, pound per second squared per foot to the fourth power

R = streamline radial location, feet

C_m = meridional velocity, feet per second

R_c = streamline radius of curvature, feet

θ = streamline angle (angle that the streamline makes with the centerline of engine), degrees

Blade losses were estimated from test data of compressors that had been designed at similar Mach number levels but at lower pressure ratio levels. Radial flows and streamline curvatures were accounted for, but these effects did not make a significant change in the air velocities because the flow path and inlet configuration were purposely designed with as small a wall curvature as possible. Figure 1 presents the axial velocity at station 1. Figure 2 shows that the streamline curvature and radial flow effects cause a very small change in axial velocity.

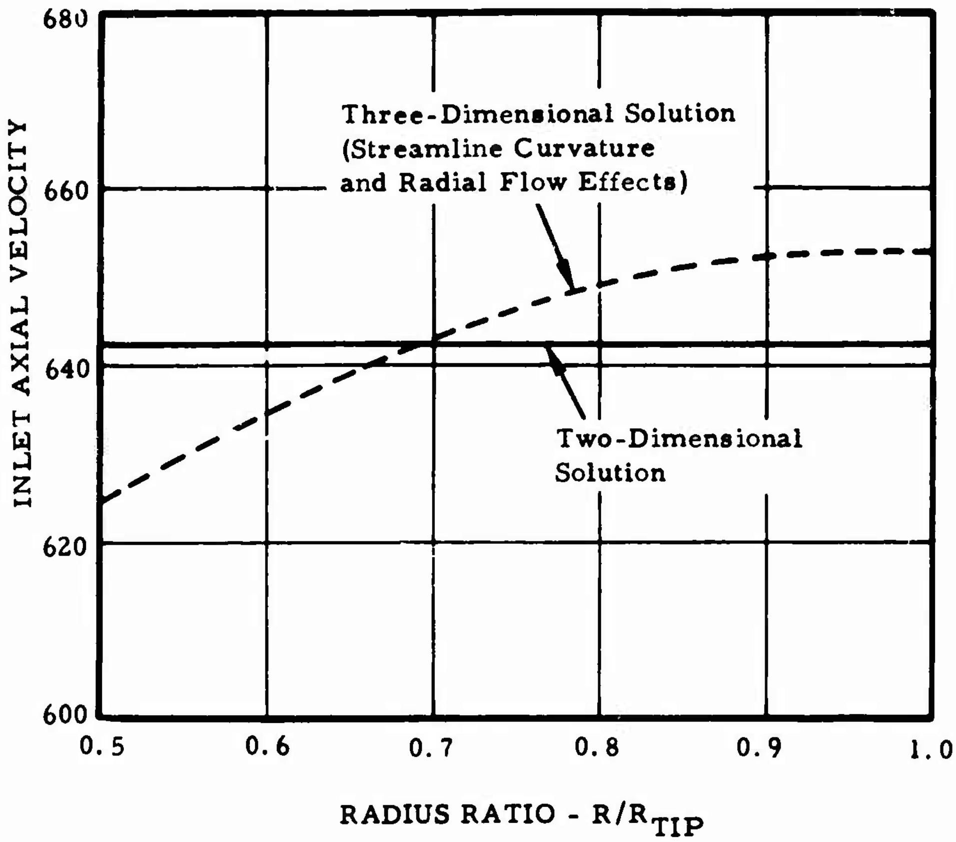


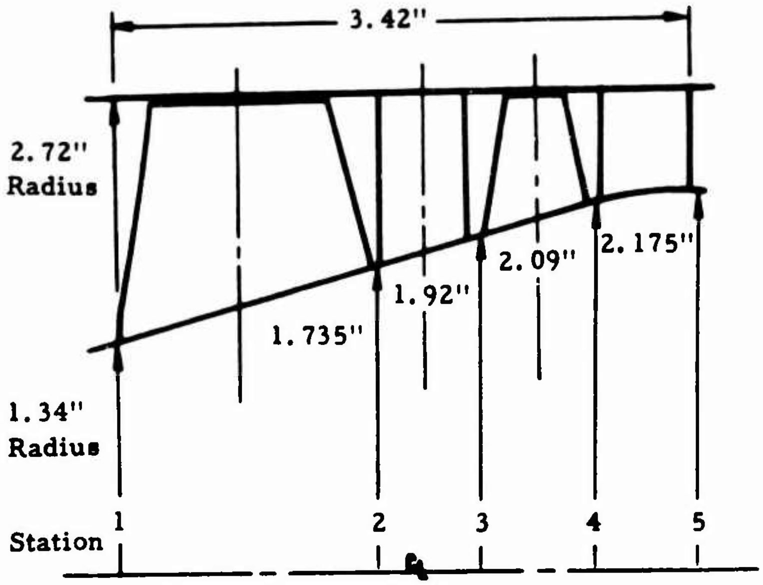
Figure 1. (C) Advanced Axial Compressor Inlet Velocity Profile. (U)

(C) DESIGN PERFORMANCE (U)

(U) The design performance of the axial compressor quoted at the end of the transition duct is listed below. A 1-percent transition duct total pressure loss was assumed.

Overall pressure ratio	3.0:1
Overall efficiency	82.3 percent
First-Stage pressure ratio	1.825:1
Second-Stage pressure ratio	1.660:1
First-Stage efficiency	85.5 percent
Second-Stage efficiency	82.9 percent
First rotor tip speed	1415 feet per second

(The slight increase in tip speed over that of the cursory design reflects an additional optimization of tip speed with rotor diffusion.)



Airflow = 5 Lb/Sec
Pressure Ratio = 3.0:1
Efficiency = 82.3 Percent
Airflow/Annulus Area = 40.96 Lb/Sec/Ft^2
Airflow/Frontal Area = 30.98 Lb/Sec/Ft^2
Inlet Hub/Tip Ratio = 0.494

Figure 2. (C) Advanced Axial Compressor Aerodynamic Flow Path. (U)

(C) The pressure ratio split between the two stages balances the roto diffusion. Because of the increase in temperature after the first st . the second-stage pressure ratio must be reduced to prevent ex- c ssive rotor diffusion.

(C) A constant radial pressure ratio distribution was used to maintain a low radial gradient of axial velocity throughout the axial compressor and into the centrifugal compressor inducer. A representative axial velocity gradient is shown in Figure 3. A low axial velocity gradient tends to spread the compressor work over the entire blade span rather than to concentrate the work in one area. Thus, the stage matching problems are minimized.

(U) The radial gradient of efficiency shown in Figures 4 and 5 reflects the relatively high shock losses at the tip region of the rotors. These losses have been minimized, however, by using a high Mach number rotor blade section that efficiently converts the kinetic energy of the air stream to static pressure.

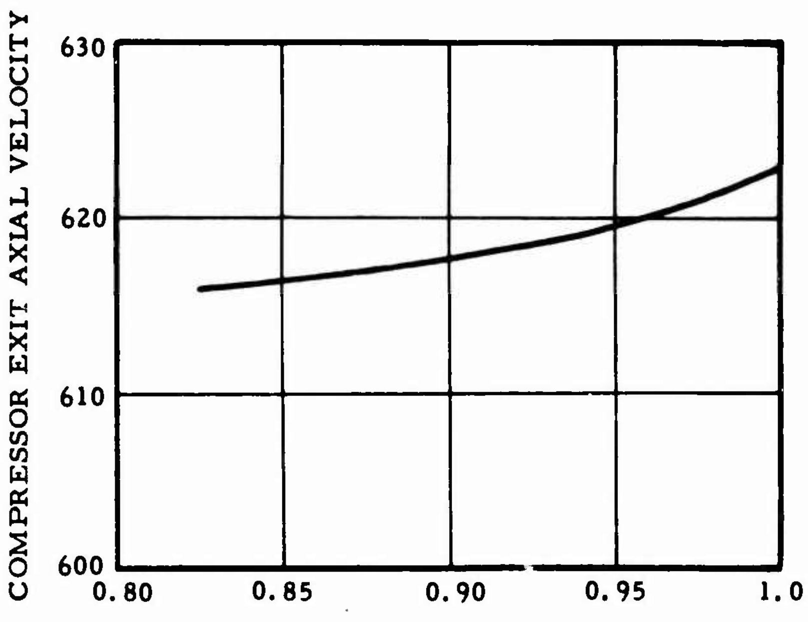
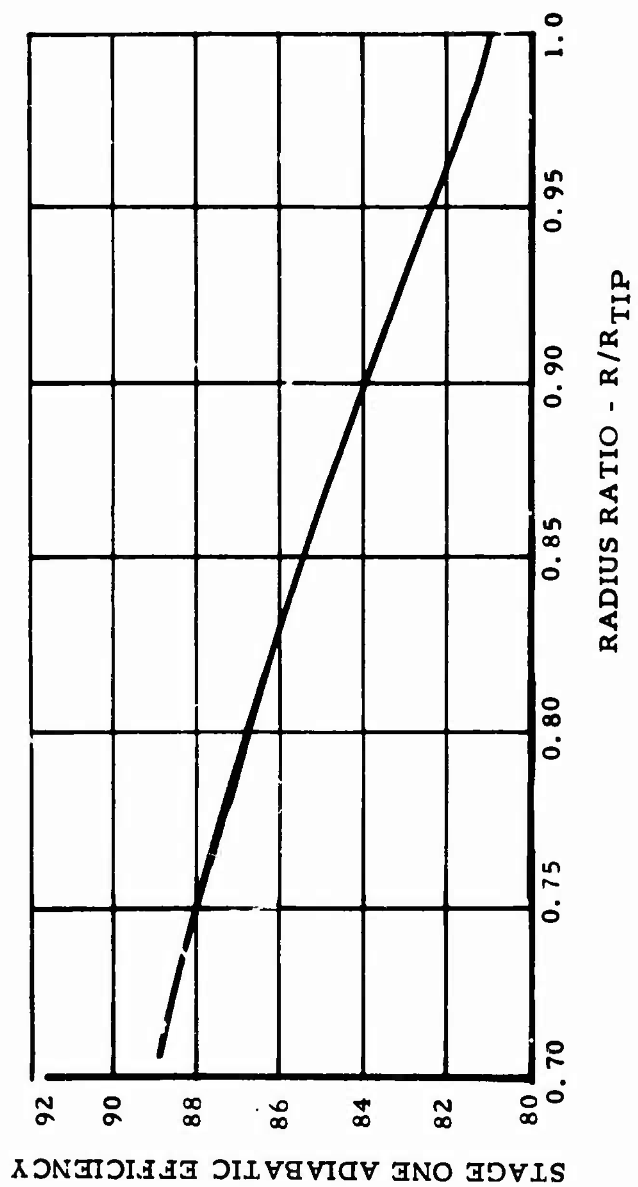


Figure 3. (C) Advanced Axial Compressor Radial Variation of Exit Velocity (Station 5). (U)



Fi_:re 4. (C) Advanced Axial Compressor Stage One - Radial Variation of Efficiency (Station 3). (U)

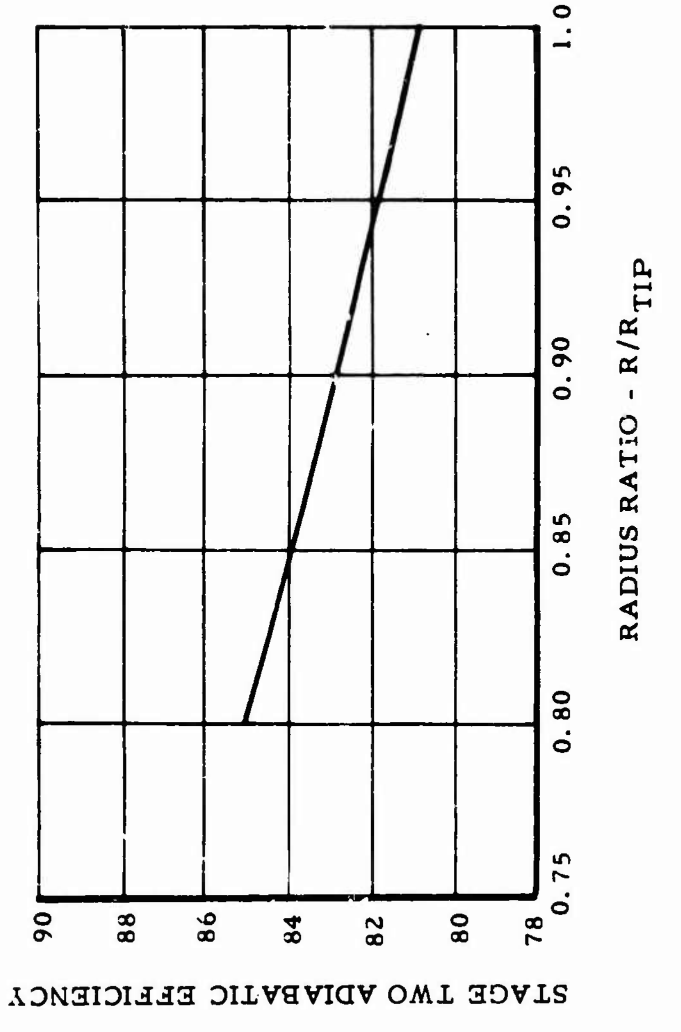


Figure 5. (C) Advanced Axial Compressor Stage Two Radial Variation of Efficiency (Station 5). (U)

(C) FLOW PATH (U)

(U) The axial compressor flow path is shown in Figure 2. This flow path embodies aspect ratio (blade height divided by the mean radius chord length) and ramp angle optimizations. In addition, aerodynamic blockage factors (flow area divided by the actual or physical area) have been included.

(C) Continental's axial compressor experience has shown that excessively low aspect blading produces high losses, and that excessively high aspect ratio blading limits flow range. The axial compressor aspect ratios, which have been optimized for efficiency and flow range, and the number of blades and the mean chords are listed in Table I.

(C) TABLE I (U)
BLADE AND MEAN CHORDS (U)

Blade Row	Aspect Ratio	Mean Chord	Number of Blades
Rotor 1	0.646	1.83 in.	15
Stator 1	1.623	0.55 in.	47
Rotor 2	0.953	0.75 in.	41
Stator 2	1.031	0.57 in.	53

(C) Blockage factors, which account for the wall boundary layers, follow in Table II. These values were obtained from test data on similar, but lower pressure ratio level, transonic axial compressors. The axial station locations are indicated in Figure 2.

(C) TABLE II (U)
BLOCKAGE FACTORS (U)

Axial Station	Blockage Factor
1	0.99
2	0.98
3	0.97
4	0.97
5	0.97

(U) It should be noted that the optimum axial compressor is about 0.47 inch longer than that of the cursory design. This is because the cursory design aspect ratios were assumed to be 1.0 .

(C) VELOCITY TRIANGLES (U)

(U) The velocity triangles, which completely define the aerodynamics of the compressor, are shown in Figures 6 through 9.

(U) Each figure presents the inlet and exit velocity triangles for the respective blade row. In addition, Mach numbers, air angles, and radii are listed. Five velocity triangles are shown for each blade row. These triangles are located at selected streamlines that enclose the percent annulus area as shown in Table III.

(U) TABLE III
STREAMLINES ENCLOSING PERCENT ANNULUS AREA

Streamline	Percent Annulus Area
11	100
9	80
6	50
3	20
1	0

Thus, for example, streamline 11 is at the aerodynamic blade tip, streamline 6 is located at the mean area location, and streamline 1 is at the aerodynamic blade hub.

(U) BLADE AND VANE GEOMETRY

The blade and vane geometry parameters listed in Tables IV and V completely define the geometric profiles of each blade row.

Circular arc vane profiles were selected for the stators because of their relatively high subsonic Mach number and air turning angle requirements. Circular arc blade profiles were also selected for the second rotor because the relative Mach numbers entering this blade row are low enough to allow the use of this profile and still maintain high efficiencies.

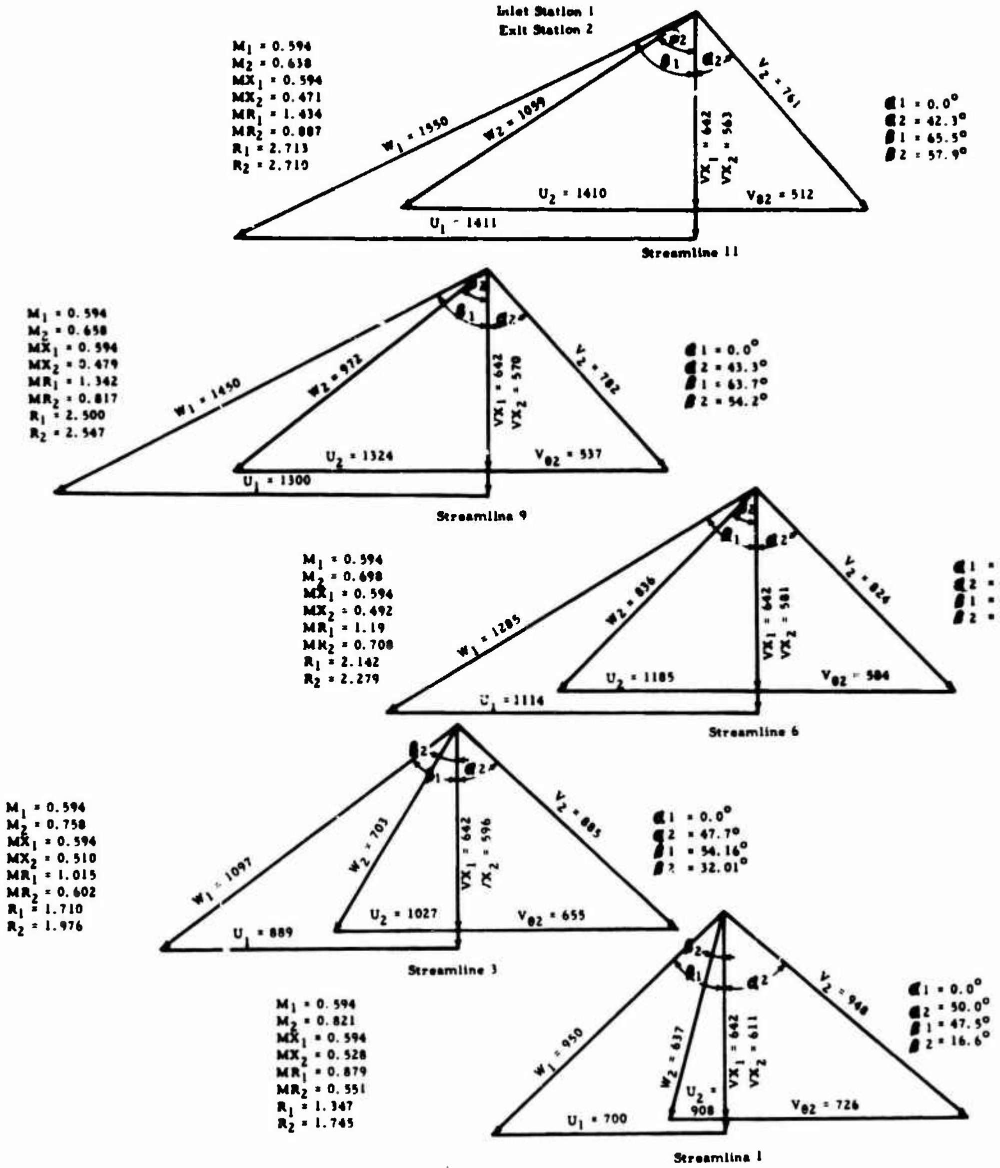


Figure 6. (C) Rotor 1 Velocity Triangles. (U)

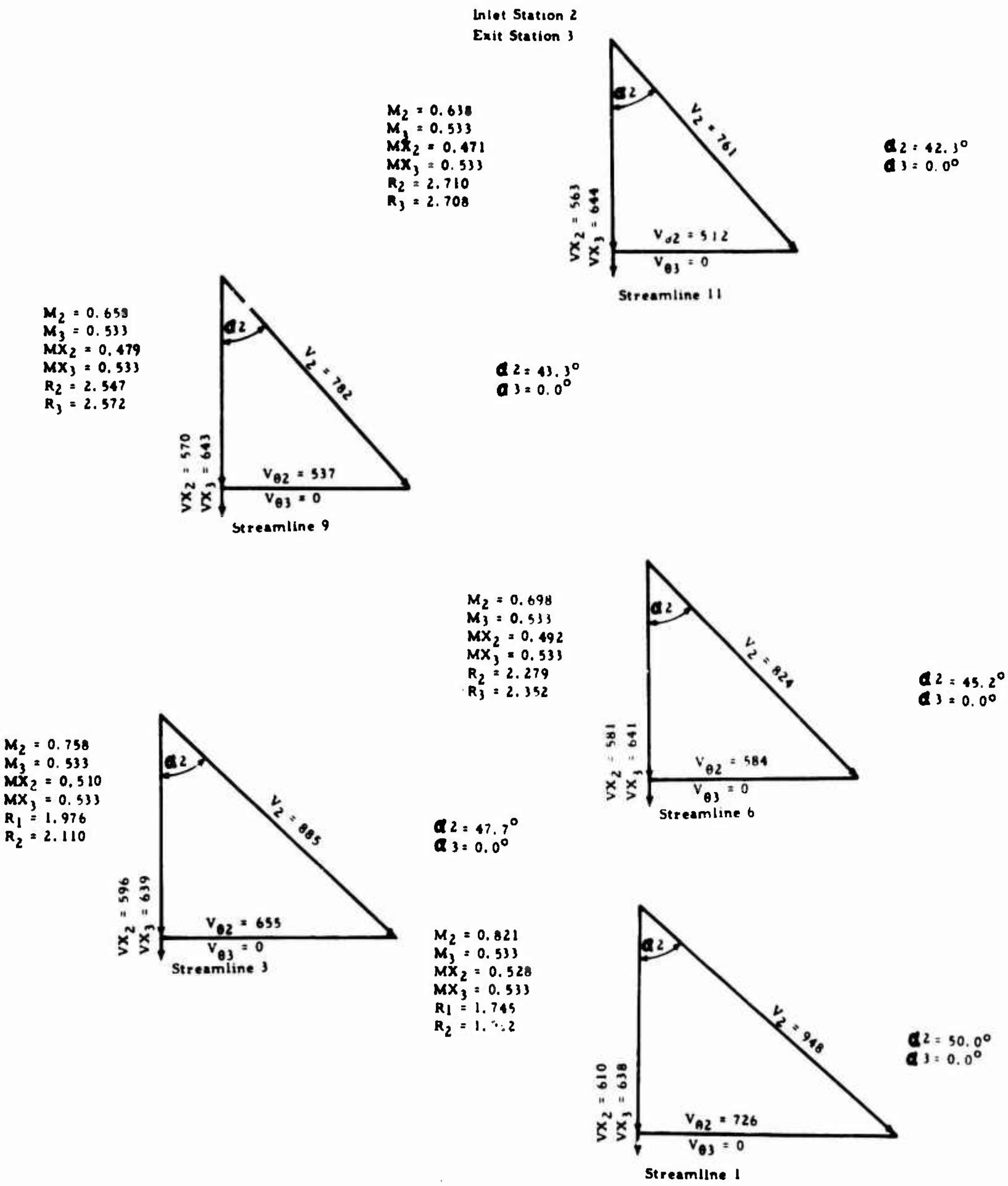


Figure 7. (C) Stator 1 Velocity Triangles. (U)

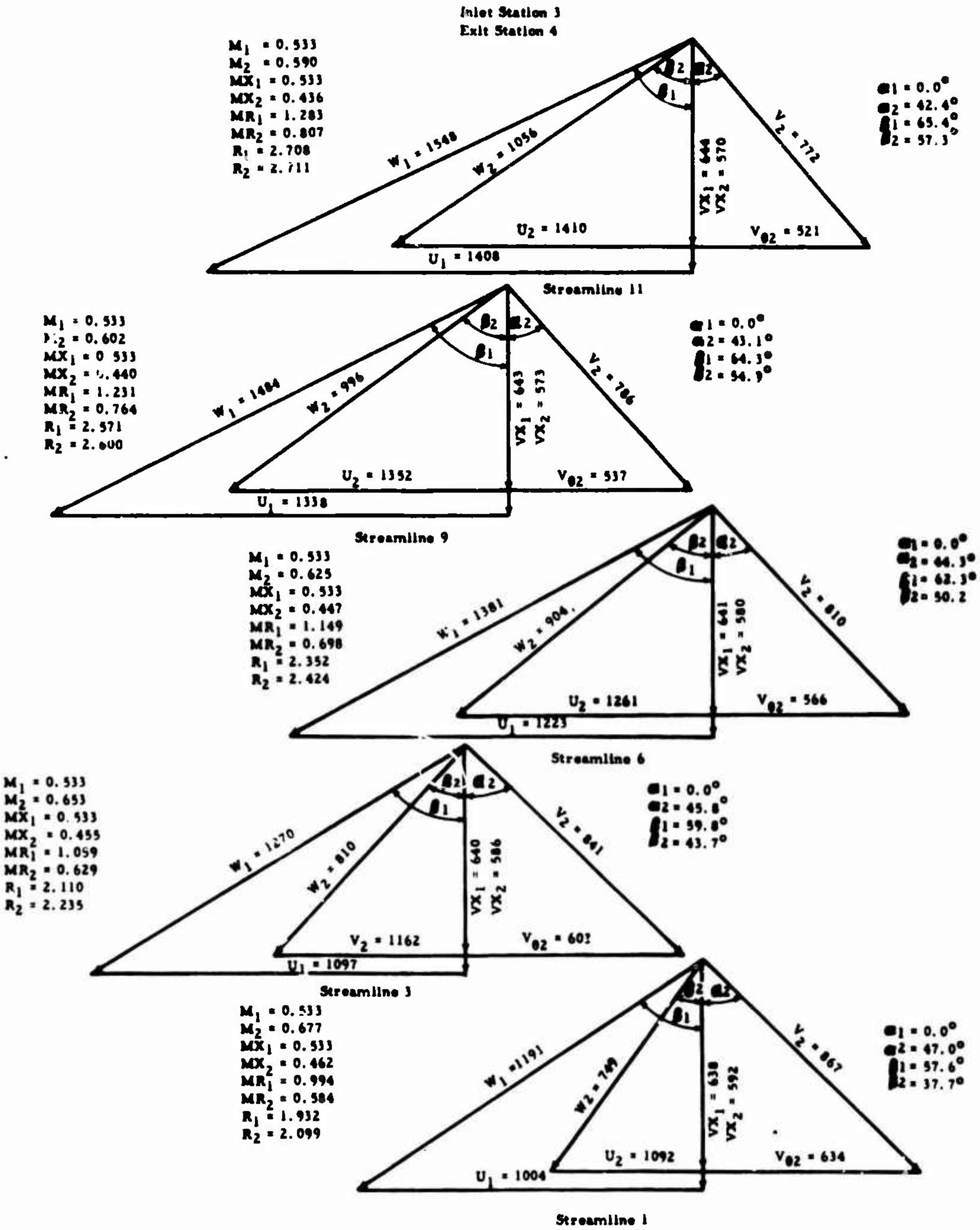


Figure 8. (C) Rotor 2 Velocity Triangles. (U)

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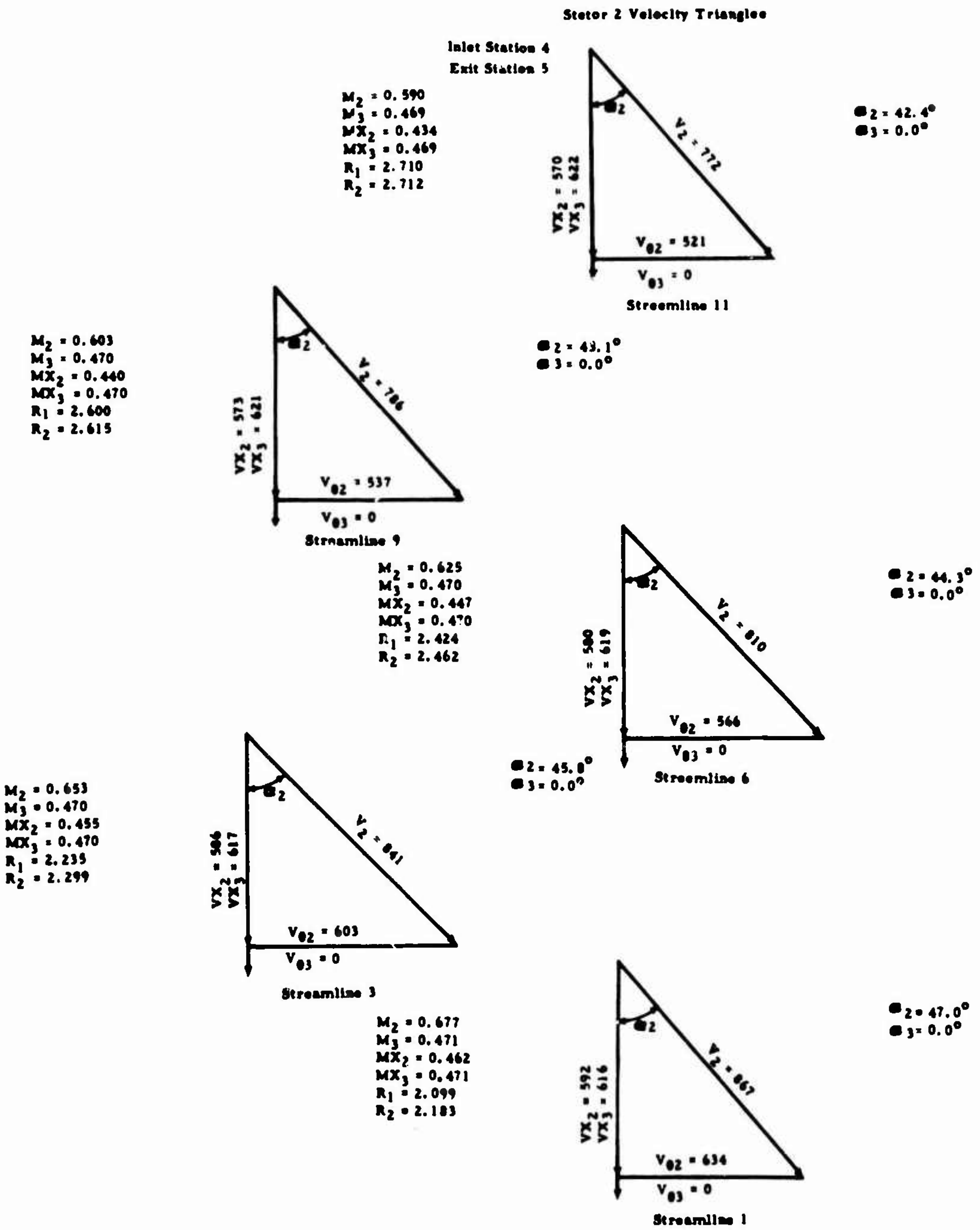

Figure 9. (C) Stator 2 Velocity Triangles. (U)

(C) TABLE IV (U)
FIRST-STAGE BLADE AND VANE GEOMETRY (U)

Type - Rotor
Profile - Linear - Double Circular Arc
Number of Blades - 15

Radius, Inches	Chord, Inches	Solidity	Maximum Thickness % - Chord	Suction Surface Radius, Inches	Pressure Surface Radius, Inches	Axial Length, Inches	Leading Edge Radius, % - Chord	Distance To Circular Arc % - Chord	Inlet Metal Angle, Degrees	Exit Metal Angle, Degrees	Camber Angle, Degrees	Turning Angle, Degrees	Deviation Angle, Degrees	Incidence Angle, Degrees	Setting Angle, Degrees
2.7200	2.050	1.799	0.0300	NA	NA	1.004	0.00195	0.560	62.60	53.87	8.72	7.60	4.12	3.0	60.68
2.4243	1.940	1.910	0.0340	NA	NA	1.088	0.00206	0.370	60.70	45.38	15.32	12.90	5.42	3.0	55.88
2.1287	1.830	2.052	0.0410	NA	NA	1.245	0.00219	0.185	56.90	32.90	24.00	20.20	6.80	3.0	47.14
1.8331	1.715	2.234	0.0543	2.048	3.569	1.424	0.00271	NA	53.10	14.61	38.49	33.30	8.19	3.0	33.86
1.5375	1.600	2.484	0.0800	1.372	2.356	1.496	0.00400	NA	48.40	6.89	55.29	50.90	7.39	3.0	20.75

Type - Stator
Profile - Double Circular Arc
Number of Vanes - 47

Section Radius, Inches	Chord, Inches	Solidity	Maximum Thickness % - Chord	Suction Surface Radius, Inches	Pressure Surface Radius, Inches	Axial Length, Inches	Leading Edge Radius % - Chord	Distance To Circular Arc % - Chord	Inlet Metal Angle, Degrees	Exit Metal Angle, Degrees	Camber Angle, Degrees	Turning Angle, Degrees	Deviation Angle, Degrees	Incidence Angle, Degrees	Setting Angle, Degrees
2.7200	.55	1.480	0.060	0.5291	0.7391	0.530	0.00909		42.30	-10.94	53.24	42.30	10.94	0.0	15.68
2.4970	.55	1.613	0.055	0.5300	0.7052	0.5275	0.00909		43.60	-10.69	54.29	43.60	10.69	0.0	16.46
2.2738	.55	1.771	0.050	0.5279	0.6704	0.5249	0.00909		45.20	-10.46	55.66	45.20	10.46	0.0	17.37
2.0507	.55	1.964	0.045	0.5248	0.6371	0.5219	0.00909		47.00	-10.21	57.21	47.00	10.21	0.0	18.40
1.8275	.55	2.203	0.040	0.5196	0.6044	0.5182	0.00909		49.10	- 9.94	59.04	49.10	9.94	0.0	19.58

(C) TABLE V (U)
SECOND-STAGE BLADE AND VANE GEOMETRY(U)

Type - Rotor
Profile - Double Circular Arc
Number of Blades - 41

Radius, Inches	Chord, Inches	Solidity	Maximum Thickness % - Chord	Suction Surface Radius, Inches	Pressure Surface Radius, Inches	Axial Length, Inches	Leading Edge Radius, % - Chord	Distance To Circular Arc % - Chord	Inlet Metal Angle, Degrees	Exit Metal Angle, Degrees	Camber Angle, Degrees	Turning Angle, Degrees	Deviation Angle, Degrees	Incidence Angle, Degrees	Setting Angle, Degrees
2.7200	0.75	1.756	0.0300	2.993	6.552	0.395	0.0067		63.50	53.02	10.48	8.10	4.39	2.0	58.26
2.5413	0.75	1.879	0.0375	2.251	5.485	0.433	0.0067		61.50	47.98	13.52	10.60	5.46	2.5	54.74
2.3625	0.75	2.021	0.0445	1.734	4.141	0.477	0.0067		59.40	41.70	17.70	14.20	6.54	3.0	50.55
2.1838	0.75	2.186	0.0520	1.356	3.059	0.525	0.0067		57.10	33.97	23.13	19.10	7.55	3.5	45.54
2.0050	0.75	2.381	0.0600	1.083	2.286	0.577	0.0067		54.60	24.84	29.76	25.50	8.26	4.0	39.72

Type - Stator
Profile - Double Circular Arc
Number of Vanes - 53

Section Radius, Inches	Chord, Inches	Solidity	Maximum Thickness % - Chord	Suction Surface Radius, Inches	Pressure Surface Radius, Inches	Axial Length, Inches	Leading Edge Radius % - Chord	Distance To Circular Arc % - Chord	Inlet Metal Angle, Degrees	Exit Metal Angle, Degrees	Camber Angle, Degrees	Turning Angle, Degrees	Deviation Angle, Degrees	Incidence Angle, Degrees	Setting Angle, Degrees
2.7200	0.57	1.801	0.060	0.5179	0.7047	0.5421	0.0088		46.65	-10.68	57.33	46.65	10.68	0.0	17.99
2.4263	0.57	2.019	0.060	0.5432	0.7590	0.5439	0.0088		44.32	-9.46	53.79	44.32	9.46	0.0	17.43
2.1325	0.57	2.297	0.060	0.5678	0.8144	0.5451	0.0088		42.33	-8.36	50.69	42.33	8.36	0.0	16.98

The relative Mach numbers entering the first rotor, however, are high enough to cause high losses if circular arc profiles are used for this blade row. Therefore, a high Mach number section was used to provide as high a stage efficiency as possible consistent with the high pressure ratio requirement. This section limits the flow expansion on the suction surface to reduce passage shock losses. A comparison of this section with a circular arc section is shown on Figure 10.

(U) VARIABLE INLET GUIDE VANES

The variable inlet guide vanes are designed to provide sufficient flow turning to ensure adequate flow range on the combined axial-centrifugal unit at part power. These vanes, which are shown on Figure 11, are cantilevered from the tip and extend to 50 percent of the blade span. A chordal taper ratio is provided to match the rotor inlet relative angles spanwise.

The advantages of 50 percent span vanes over full span vanes are:

- Mechanical simplicity.
- Cleaner aerodynamic flow path (no hub variable inlet guide vane shaft protrusion or clearance space as the guide vanes are rotated).
- Less weight.

The minor disadvantage of the part span vanes is that the first rotor design inlet air angles cannot be exactly matched spanwise. Figure 12 shows that the maximum rotor inlet air angle difference with guide vane rotation occurs at the hub and is less than 5 degrees in magnitude. This difference would be less with full span vanes. However, no incidence angle problems will exist at the hub since the low inlet relative Mach numbers at this station will allow incidence angle variations of this magnitude without occurrence of severe losses.

In addition, Continental's test experience has shown that when full span variable guide vanes are severed 50 percent from the hub no significant change in overall compressor performance is noticed.

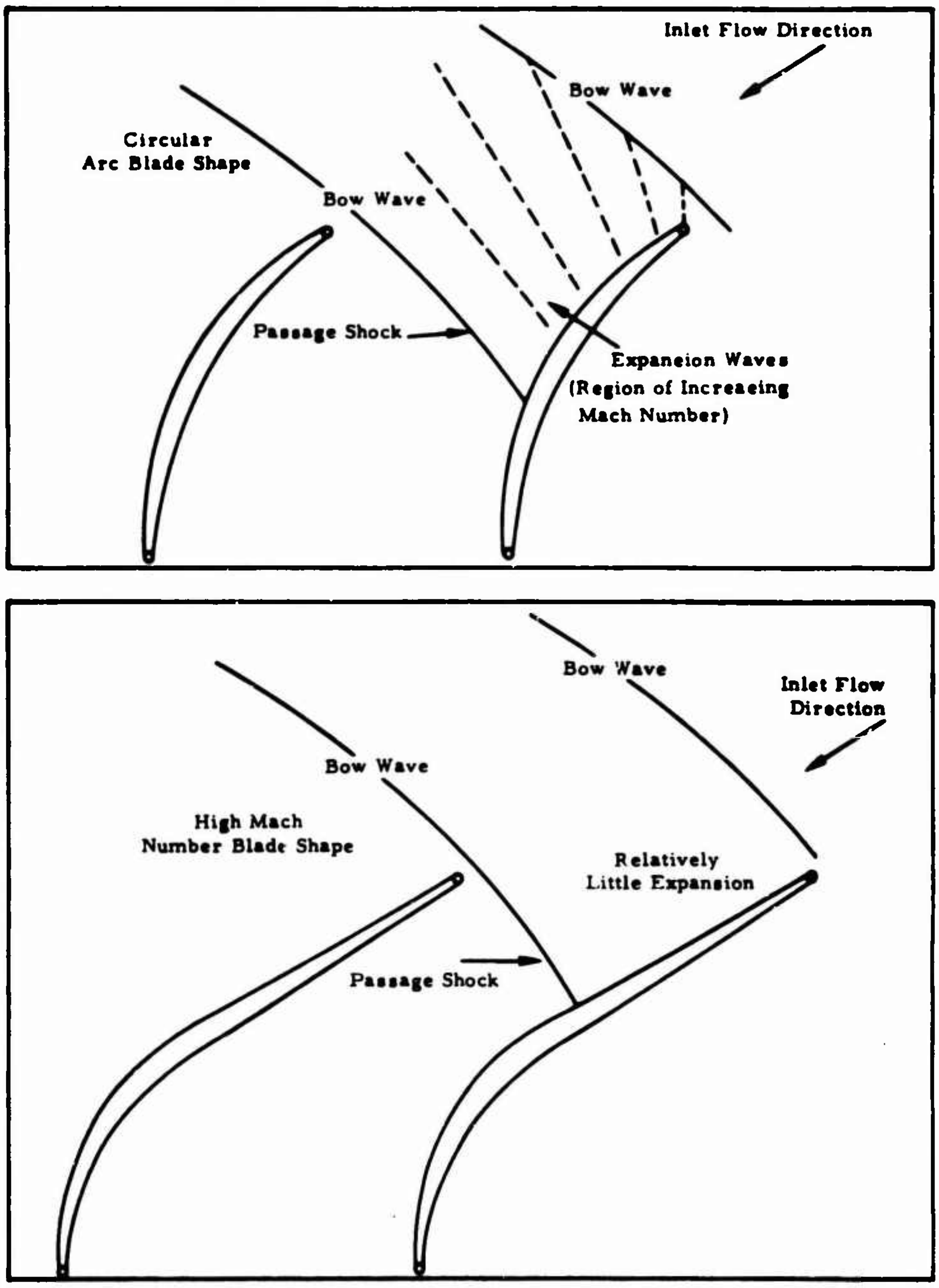

Figure 10. (U) Comparison of High Mach Number Blade Shape to Circular Arc Blade Shape.

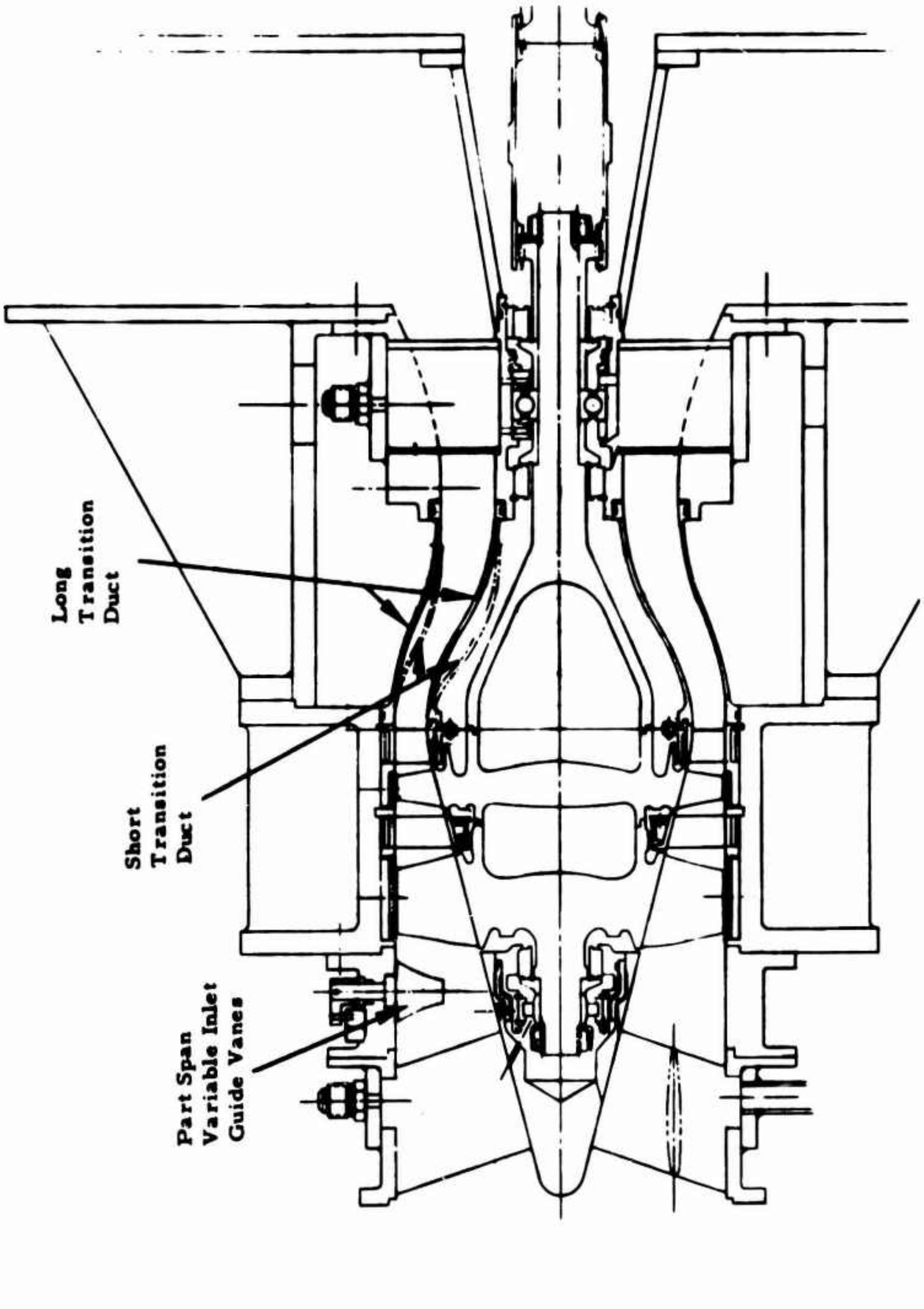

Figure 11. (U) Advanced Axial Compressor Preliminary Design Layout.

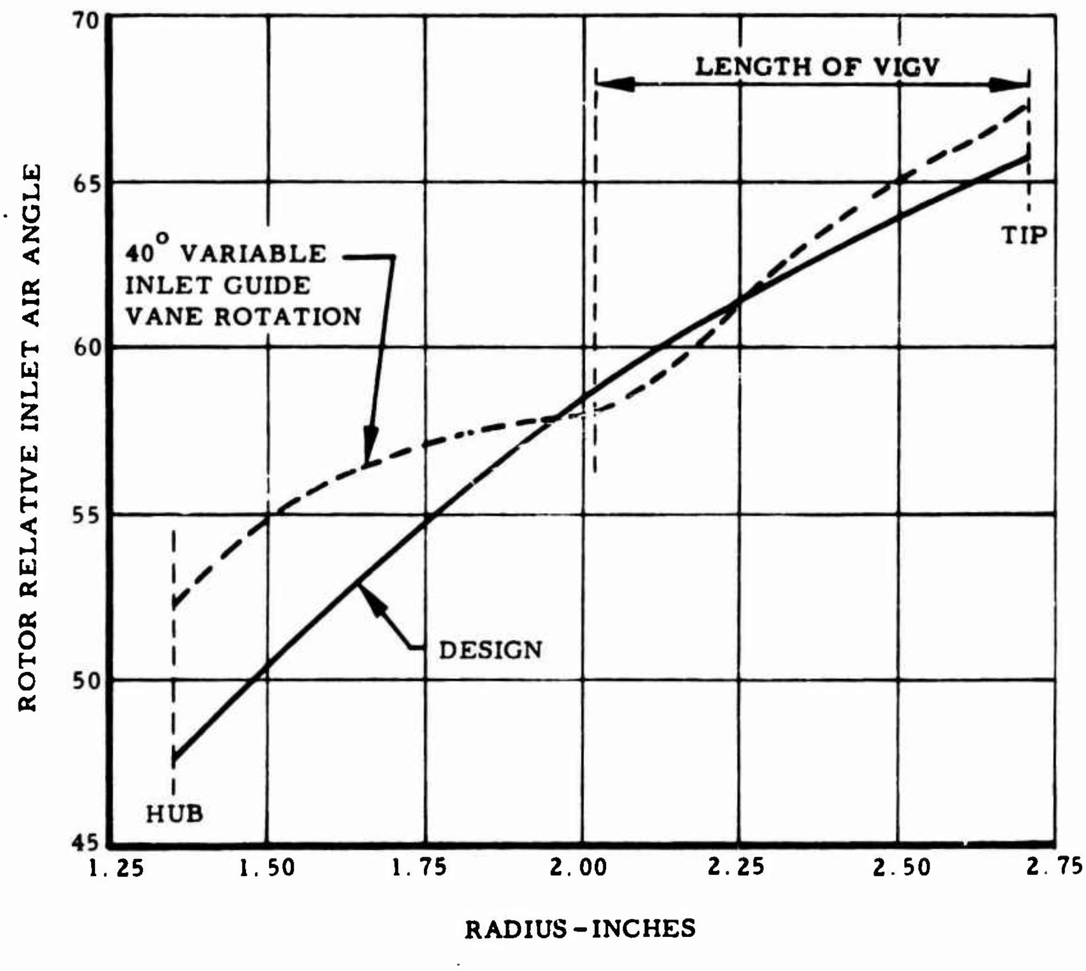

Figure 12. (U) Comparison of First Rotor Inlet Relative Air Angles With and Without Variable Guide Vane Rotation.

(U) TRANSITION DUCTS

As specified in the contract, two transition ducts were designed (see Figure 11). The long duct is 3.8 inches in length, and the short duct is 2.7 inches in length. Both ducts have the same inlet and exit radii. The velocity profiles for the long duct are presented on Figure 13, and those for the short duct are shown on Figure 14. The significant difference between the two profiles is the lower rate of velocity deceleration at the hub of the long duct at an axial station about 1.75 inches from the duct inlet plane. This region has been carefully designed to avoid flow separation. The short duct was constructed as compact as possible within boundary layer separation limits.

The wall static pressure ratios for each transition duct are shown on Figures 15 and 16. These calculated curves will be compared with the test static pressure ratio values in Phase II.

The transition ducts were designed on Program 08.047. This program predicts transition duct performance by solving the continuity, energy, and complete radial equilibrium equations throughout the flow passage. The radial equilibrium equation used is the same as that shown under Design Procedure, page 11. Radial flows and streamline curvatures were accounted for.

(U) ESTIMATED PERFORMANCE

The estimated performance map for the advanced axial compressor is presented on Figure 17. These performance data were generated from stage characteristics of similar transonic axial compressors. The compressor map is similar to the 3.1:1 pressure ratio cursory design axial compressor map and should exhibit equivalent axial-centrifugal compressor performance when matched to USAAVLABS centrifugal compressor technology.

The estimated performance map for the advanced axial compressor with variable inlet guide vanes is shown on Figure 18. This is the performance that would result from rotating the variable inlet guide vanes the maximum design value of 40 degrees in the direction of rotor rotation. (For optimum overall axial-centrifugal compressor performance at part speed conditions, a much smaller variable guide vane setting angle would be required, as was shown in the cursory design study.)

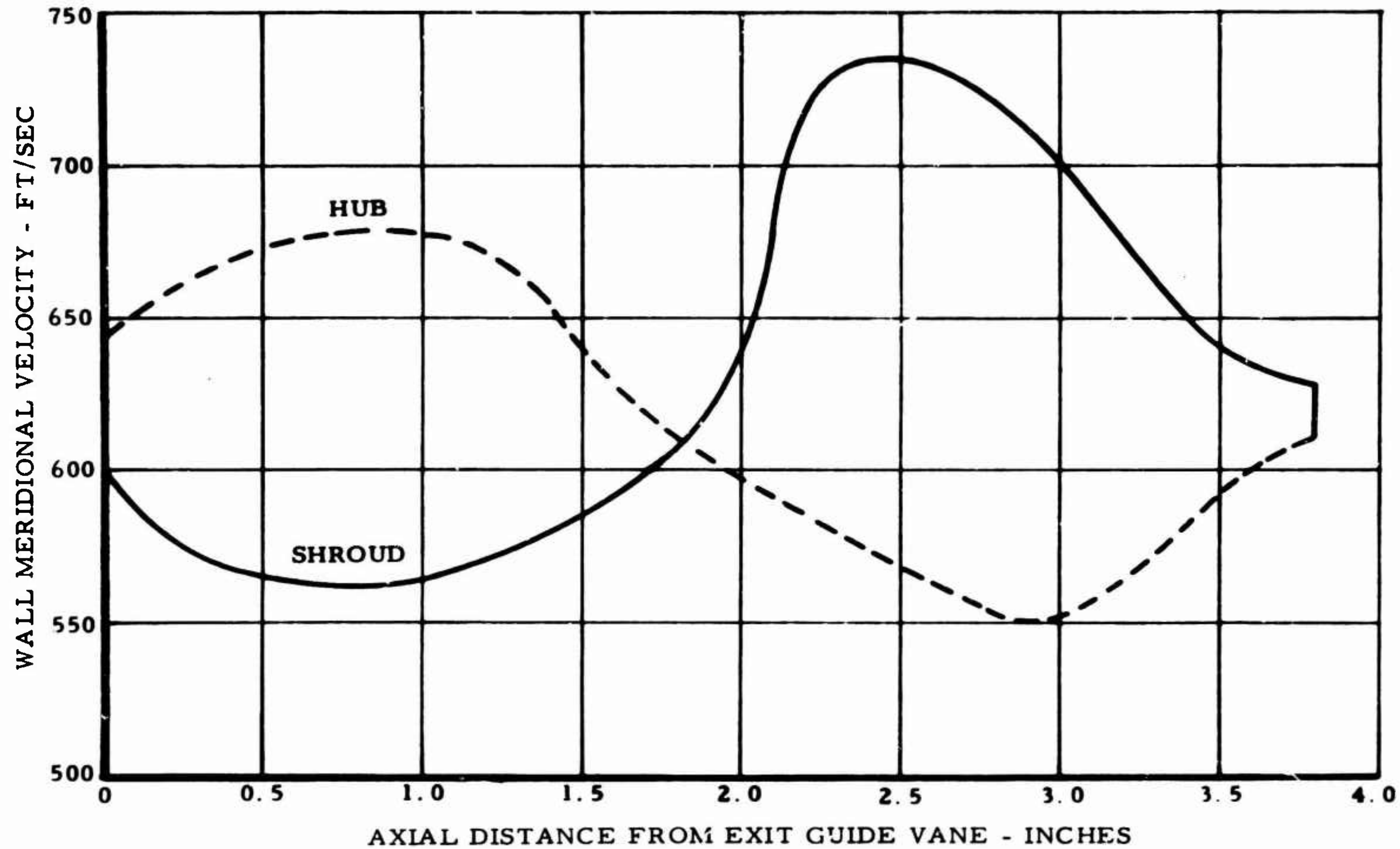

Figure 13. (U) Long Transition Duct Wall Velocity Profile.

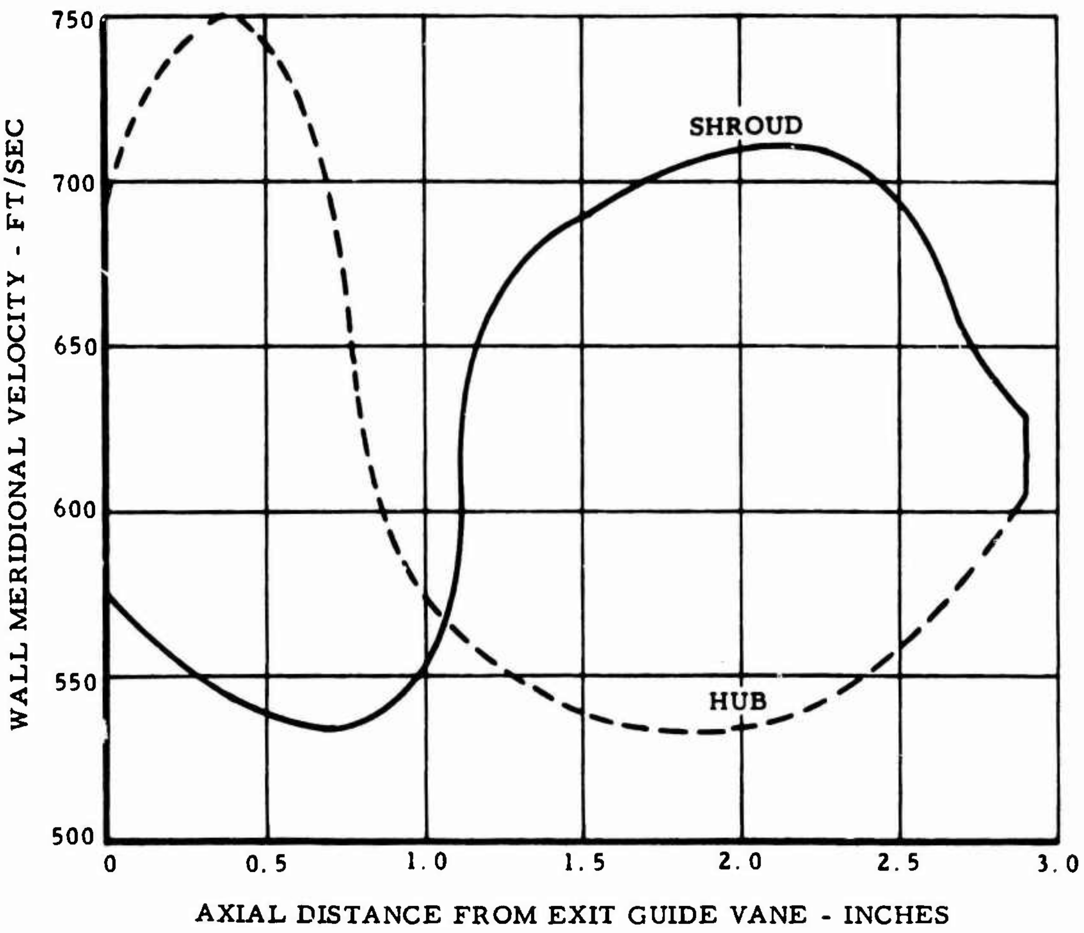

Figure 14. (U) Short Transition Duct Wall Velocity Profile.

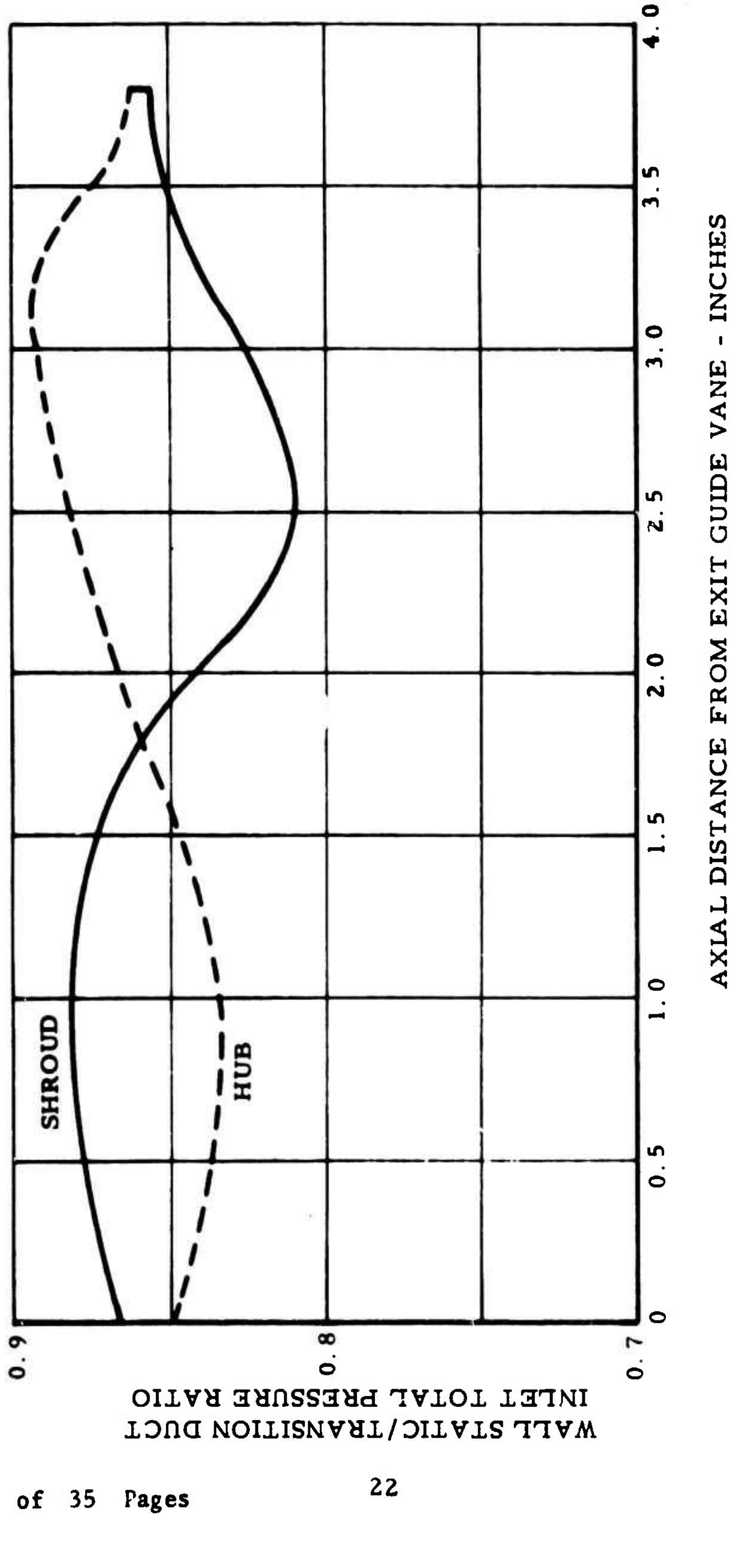

Figure 15. (U) Long Transition Duct Wall Static/Total Pressure Ratio Profile.

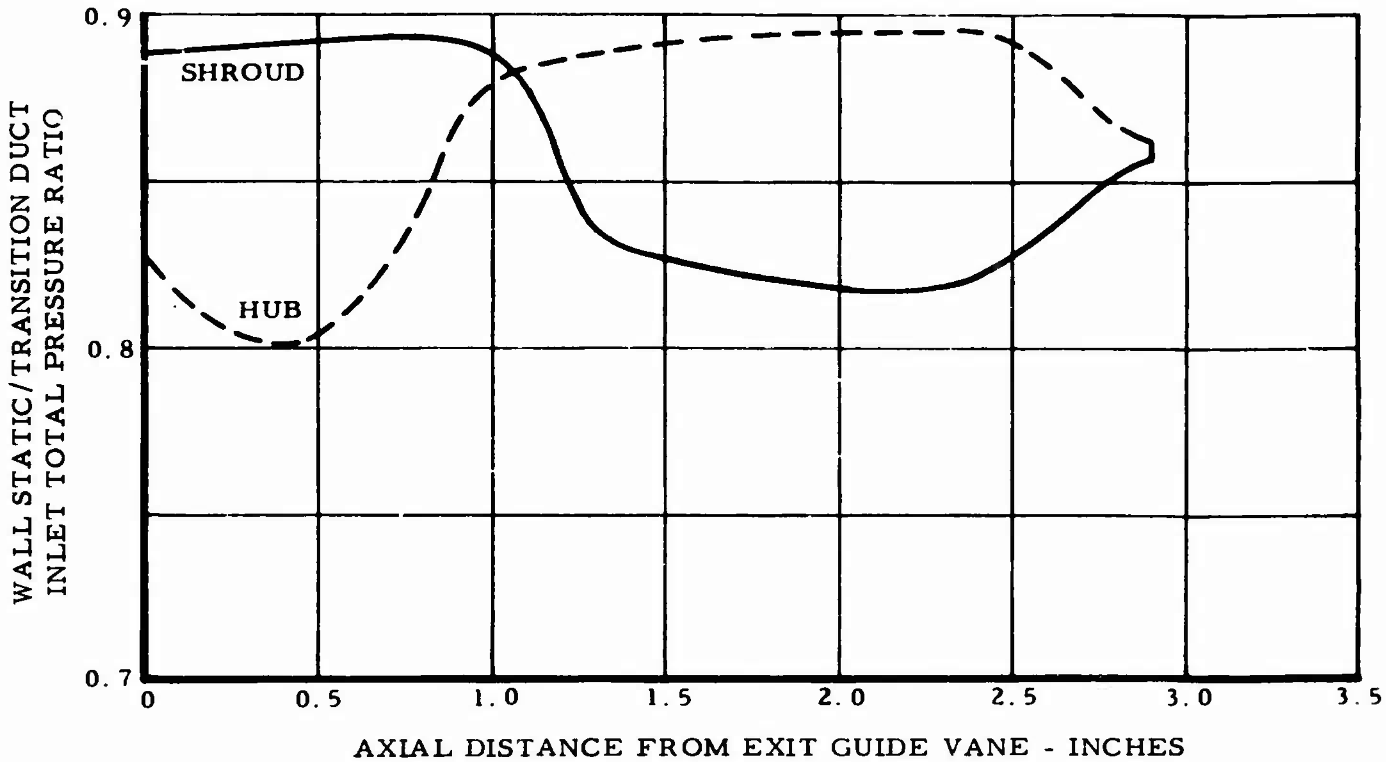

Figure 16. (U) Short Transition Duct Wall Static/Total Pressure Ratio Profile.

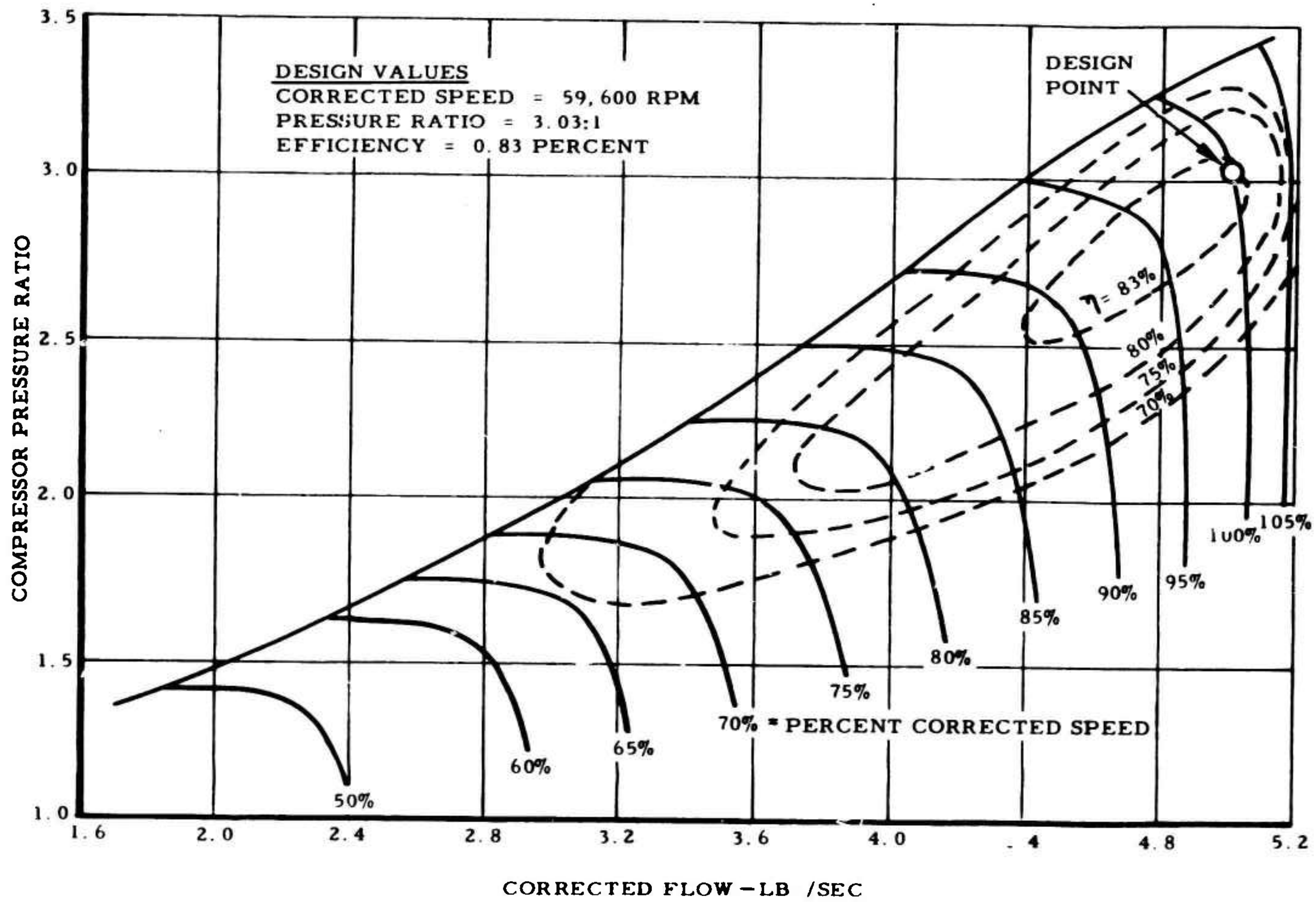

Figure 17. (U) Advanced Axial Compressor Predicted Performance Map.

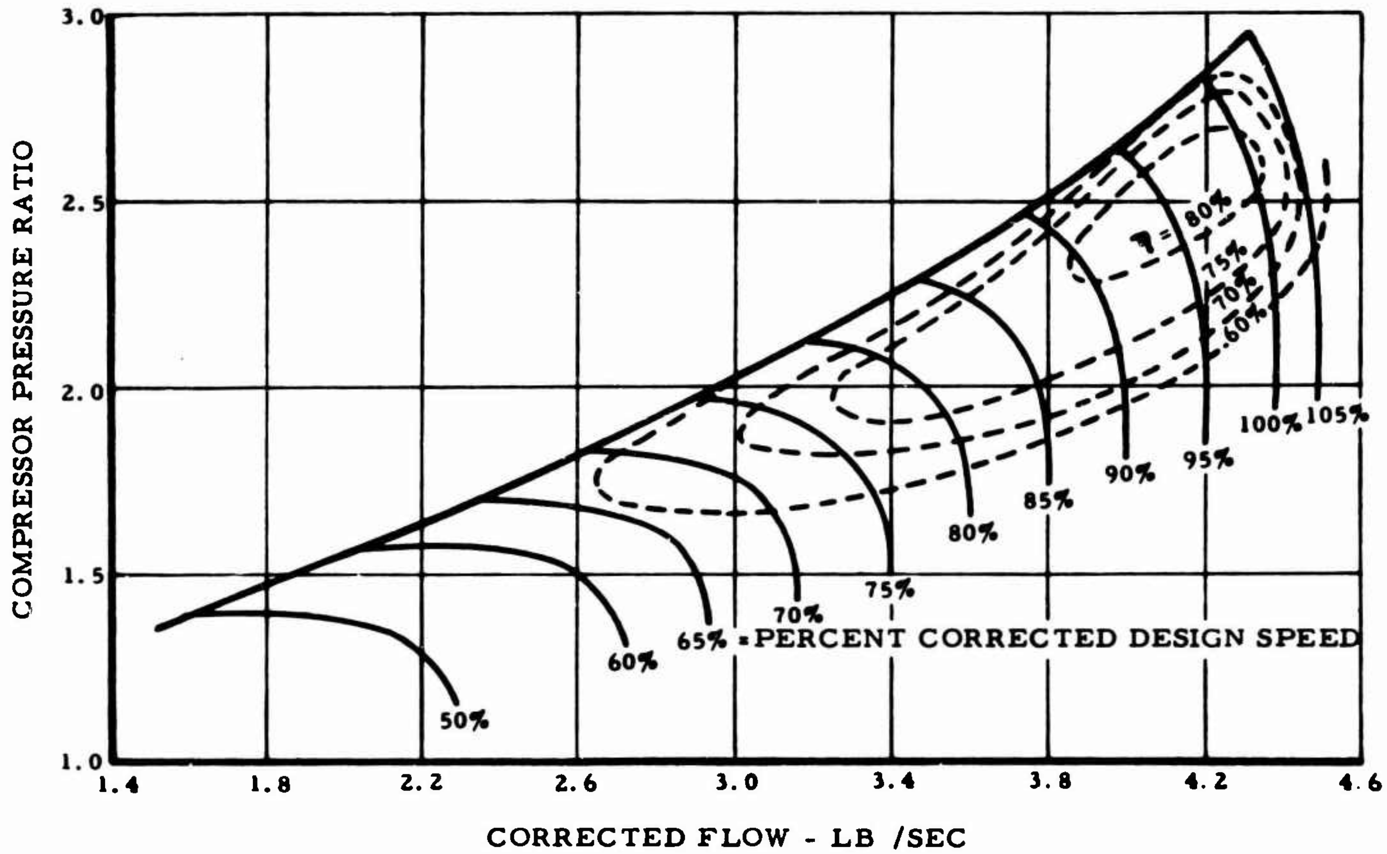

Figure 18. (U) Advanced Axial Compressor Predicted Performance Map With 40-Degree Variable Inlet Guide Vane Rotation in the Direction of Rotor Rotation.

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13. ABSTRACT

(U) A complete two-stage aerodynamic axial compressor design capable of being matched with the USAAVLABS advanced centrifugal technology was prepared. Fully variable part span inlet guide vanes were designed. Two transition ducts were also designed to provide an efficient aerodynamic flow path from the axial compressor exit to the centrifugal compressor inlet.

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